MULTIMEDIA UNIVERSITY

STUDENT ID NO

MULTIMEDIA UNIVERSITY

FINAL EXAMINATION

TRIMESTER 2, 2018/2019

EME4026 – TRIBOLOGY

(ME)

4 MARCH 2019
9:00 a.m. - 11:00 a.m.
(2 Hours)

INSTRUCTIONS TO STUDENT

1. This Question paper consists of 14 pages including cover page with **4 Questions** only.

2. Attempt **FOUR** out of **FOUR** questions. All questions carry equal marks and the distribution of the marks for each question is given.

3. Please write all your answers in the Answer Booklet provided.

4. A list of useful equations and charts are given as **Appendix A**.

Question 1

(a) **Differentiate** between the mechanisms of adhesion friction in sliding motion and adhesion friction in rolling friction. Illustrate with neat sketch.

[8 marks]

(b) In a linear scratch experiment, two diamond conical indenters are scratched along a metal surface. The indenters have semi-angle values of 65° and 75° respectively. The recorded coefficient of friction values are 1.20. The sliding mechanism is predominantly contributed by adhesion and ploughing and these are additive.

i. **Derive** the equation for ploughing component of coefficient of friction. Illustrate with neat sketch.

[11 marks]

ii. **Determine** the adhesive component of coefficient of friction for each indenter.

[6 marks]

Continued...

Question 2

In a pin-on-disc setting shown in **Figure Q2**, a solid brass bar with flat square end of cross section 15 x 15 mm^2 is subjected to a 15 N normal load and rubs against a smooth steel disc of 100 mm diameter. The bar is located at 40 mm distance from the disc center. The disc rotates about its center at 100 rpm for 150 hours. The results of the sliding were 50 mg and 5 mg mass lossess of the brass pin and steel disc respectively.

Given: Hardness of brass = 0.8 Gpa
Hardness of steel = 2.5 Gpa
Density of brass = 8.5 Mg/m^3
Density of steel = 7.8 Mg/m^3

i. **Determine** the adhesive wear coefficient **[13 marks]**

ii. **Determine** the wear depth of the brass bar **[5 marks]**

iii. **Determine** the wear depth of the steel disc **[7 marks]**

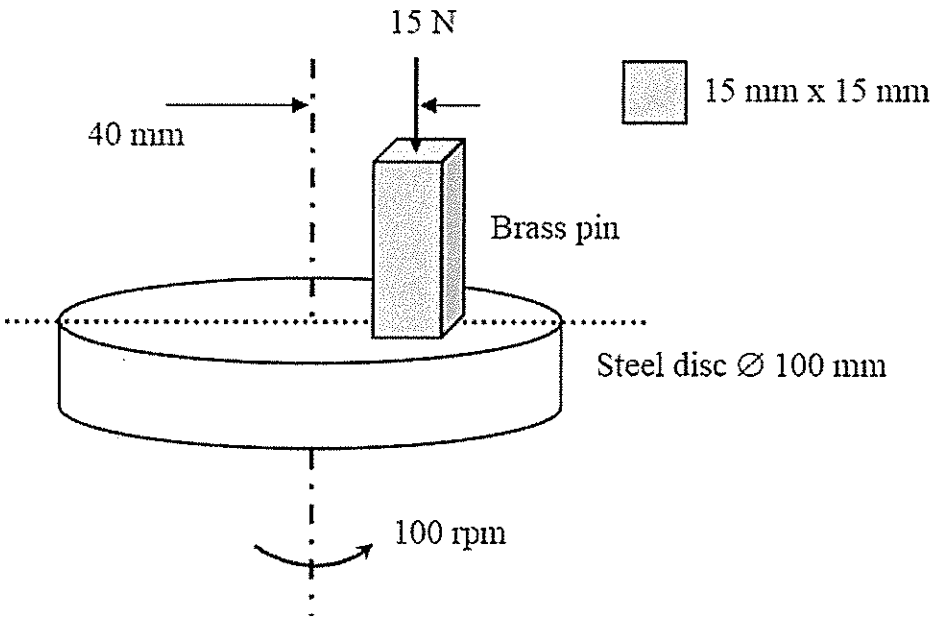

Figure Q2

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Question 3

A journal bearing of diameter, D = 50.8 mm, length of L = 25.4 mm and a radial clearance of 0.0254 mm supports a 1780 N load while rotating at 1800 rpm. SAE 10 oil is used, with an average temperature of 130°F (~55°C). **Estimate** the following:

i. Sommerfeld number **[5 marks]**

ii. Minimum oil film thickness, h_o **[2 marks]**

iii. Coefficient of friction, f **[3 marks]**

iv. Maximum film pressure, P_{max} **[3 marks]**

v. Angle between load direction and minimum film thickness, Φ **[2 marks]**

vi. Angle between load direction and termination of film, Θ_{po} **[2 marks]**

vii. Angle between load direction and maximum film pressure, Θ_{pmax} **[2 marks]**

viii. Total circumferential oil flow rate, Q **[3 marks]**

ix. Side or leakage flow rate, Q_s **[3 marks]**

Continued...

Question 4

(a) A lightly loaded journal bearing has a diameter of 50 mm and bearing width of 25 mm. It operates at a speed of 1500 rpm and lubricant used is SAE 40 oil at 20°C. The bearing characteristic number is S = 0.2.

i. **Determine** the friction torque. **[7 marks]**

ii. **Determine** the power loss (hp) due to viscous dissipation. **[3 marks]**

iii. **Determine** the average film pressure. **[3 marks]**

iv. **Determine** the radial load capacity. **[2 marks]**

(b) According to a recent study, the top five reasons for premature bearing failures are caused by *improper lubrication*, *fatigue*, *incorrect mounting*, *improper handling* and *contamination*. In your opinion, how to prevent or minimize the risk for each cause of failure?

[10 marks]

Continued...

Appendix A

- **Adhesion friction force** $F_a = A_r\,\tau_a$
- **Adhesion friction coefficient for dry contact:**

$$\mu_a = \frac{F_a}{W} = \frac{A_r.\tau_a}{W} = \frac{A_r.\tau_a}{P_r.A_r} = \frac{\tau_a}{P_r}$$

- **The friction coefficient in terms of surface roughness in elastic region,**

$$\mu_a \approx \frac{3.2(\tau_a)}{E^*\left(\frac{\sigma_p}{R_p}\right)^{1/2}}, \quad \frac{1}{E^*} = \frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}, \quad \sigma_p = \sqrt{\sigma_{p1}^2 + \sigma_{p2}^2}, \text{ and } \quad \frac{1}{R_p} = \frac{1}{R_{p1}} + \frac{1}{R_{p2}}$$

- **Degree of plasticity.** $\varphi = \left(E^*/H\right)\left(\sigma_p/R_p\right)^{1/2}$
- **Ploughing friction due to a circular cone hard asperity**

$$\mu_p = \frac{F_f}{W} = \frac{2\,d}{\pi\,r} = \frac{2\,\tan\theta}{\pi}, \quad \text{or} \quad \alpha = 90^\circ - \theta, \quad \mu_p = \frac{2\,\cot\alpha}{\pi}$$

- **Ploughing friction due to a spherical hard asperity**

$$\mu_p = \frac{F_f}{W} = \frac{P_y.A_p}{P_y.A_l} = \frac{A_p}{A_l} = \frac{2}{3\pi}.\frac{2r}{R}$$

- **Ploughing friction due to a spherical asperity for a relatively large groove width as compared to sphere radius:**

$$\mu_p = \frac{2}{\pi}\left\{\left(\frac{R}{r}\right)^2 \sin^{-1}\left(\frac{r}{R}\right) - \sqrt{\left(\frac{R}{r}\right)^2 - 1}\right\}$$

- **Ploughing friction due to a cylindrical asperity:**

  Case-1: the cylindrical placed transversely over a softer surface, $\mu_p = \sqrt{\frac{1}{2(R/d)-1}}$

  Case-2: the cylindrical placed upright over a softer surface, $\mu_p = (2/\pi)\frac{d}{R}$

- Holm equation for adhesive wear, $V_W = \frac{k_{adh}.W.x}{H}$ *Plastic contacts*
- Archard equation for adhesive wear, $V_W \alpha \frac{W.x}{H} = k_{adh}\frac{W.x}{H}$ *Plastic contacts*
- Bhushan equation for adhesive wear, $V_w = \bar{k}_{adh}\frac{W.x}{E^*\sqrt{(\sigma_p/R_p)}}$ *elastic contacts*
- Rabinowicz's equation for abrasive wear, $V_W = \frac{2W.x.\tan\theta}{\pi.H}$
- An abrasive expression similar to Archard's equation, $V_W = \frac{K_{abr}.W.x}{H}$

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Petroff's Equation

Friction torque $$T_f = \frac{4\pi^2 . \eta . n . L . R^3}{C} = f . W . R = f (D . L . P) R$$

Coefficient of friction $$f = 2\pi^2 \left(\frac{\eta n}{P}\right) \cdot \left(\frac{R}{C}\right)$$

Power loss (watt) $$H_v = 2\pi . T_{f\,(N.m)} . n_{rps}$$

Sommerfeld number $$S = \left(\frac{R}{C}\right)^2 \frac{\mu n}{P}$$

Average film pressure $$P = \frac{W}{DL}$$

Chart A1 – Variation of absolute viscosity with temperature for various lubricants

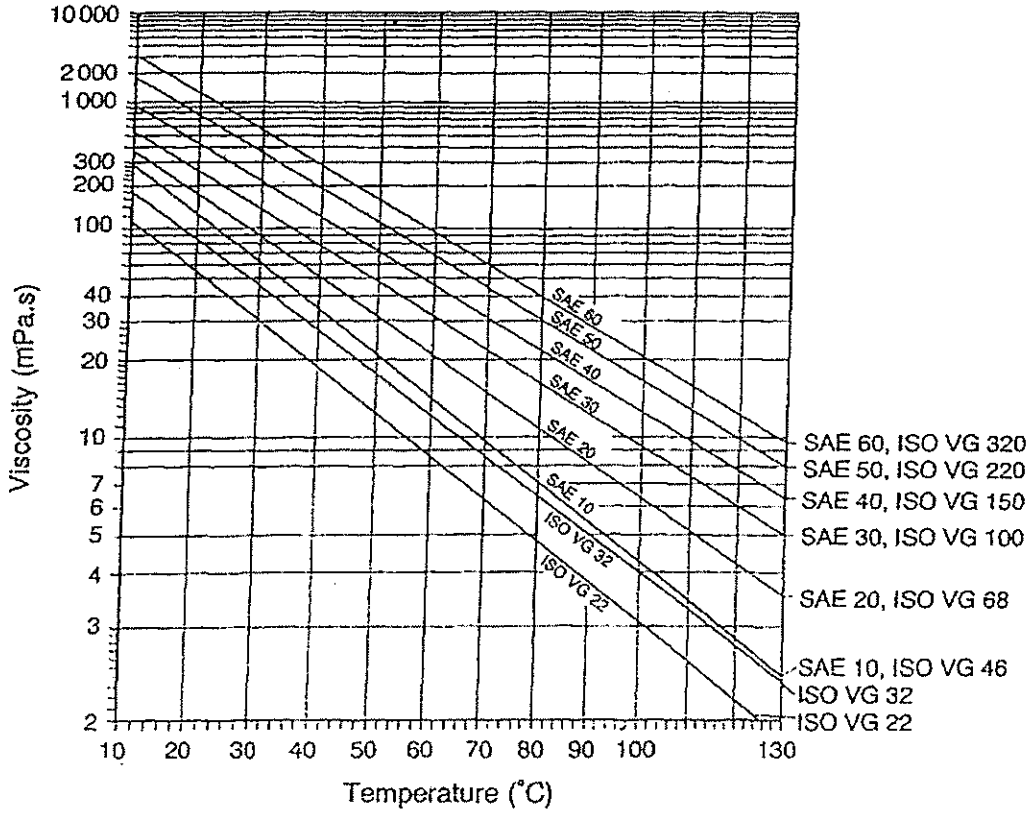

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Chart A2 – Viscosity versus temperature curves for typical SAE graded oil

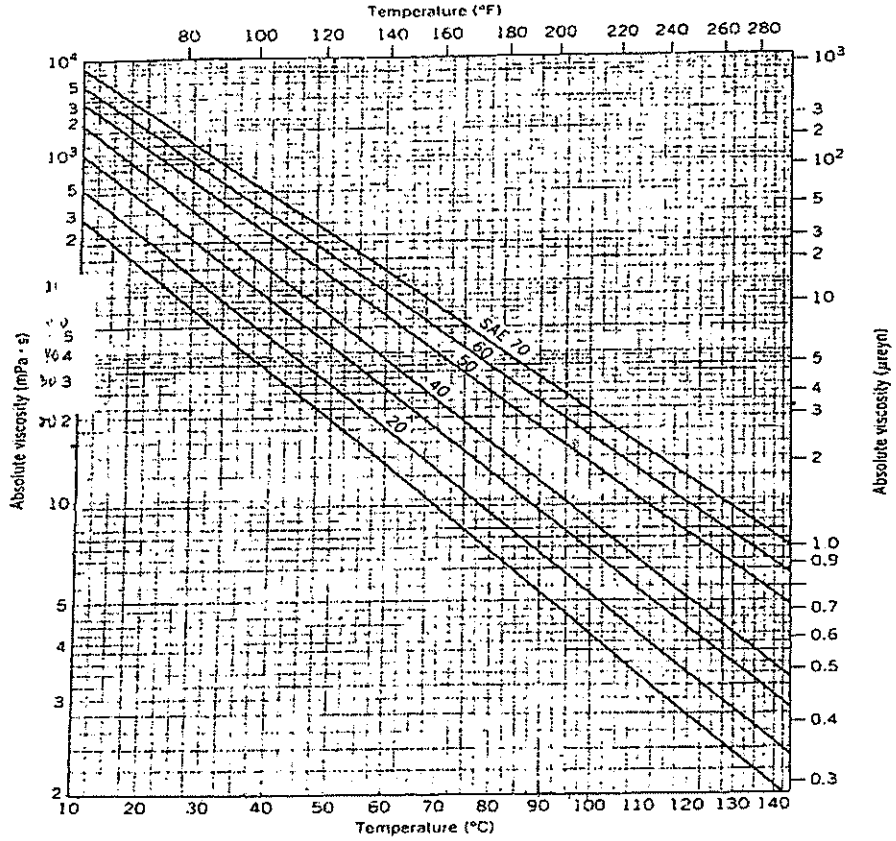

Chart A3 – Recommended values of diametral clearance (2c) for steadily loaded journal bearing

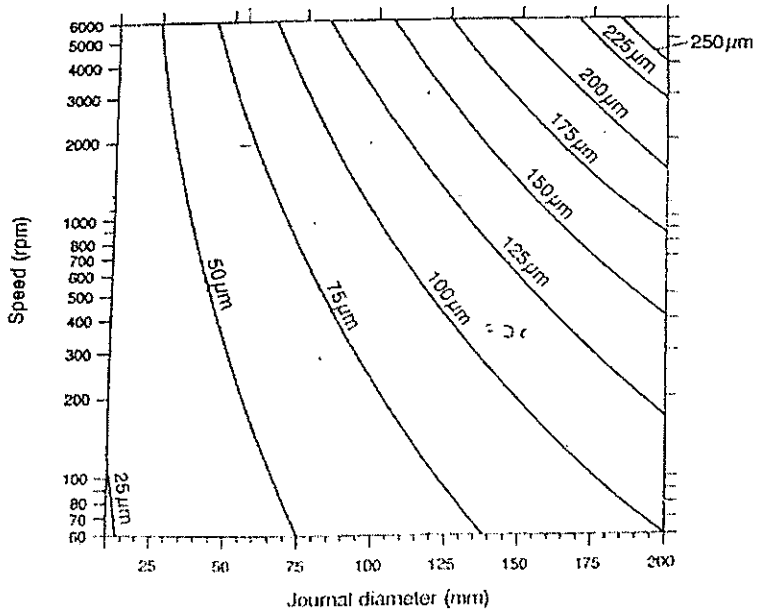

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Chart A4 – Minimum film thickness variable

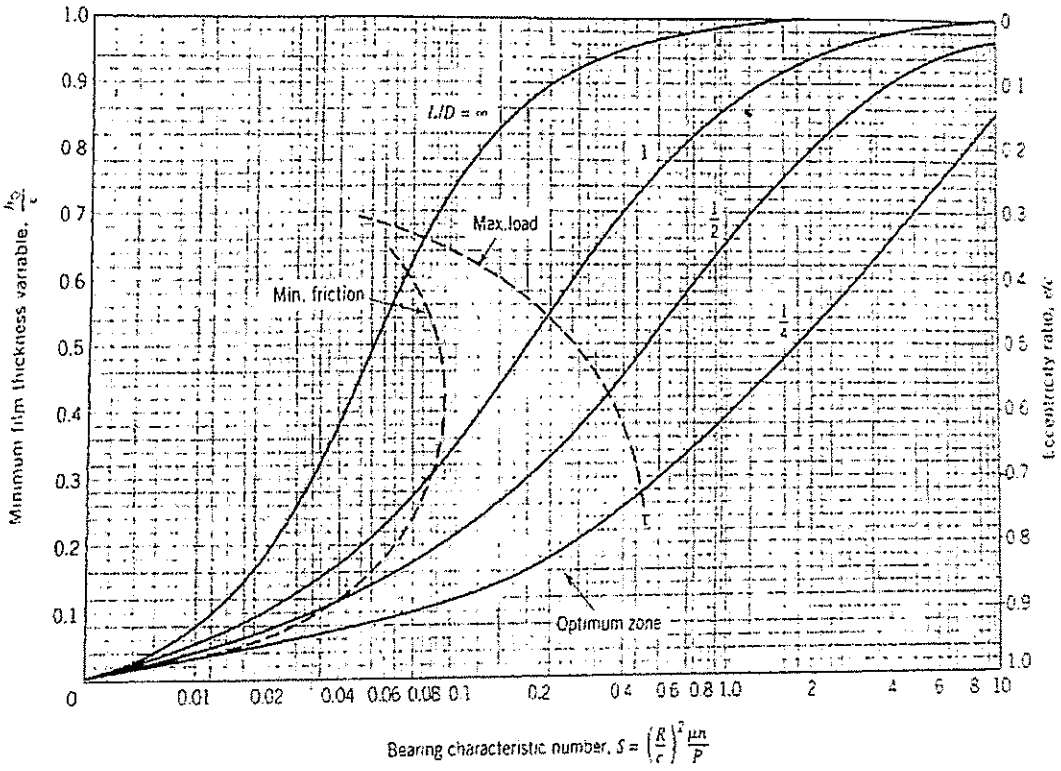

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Chart A5 – Friction coefficient variable

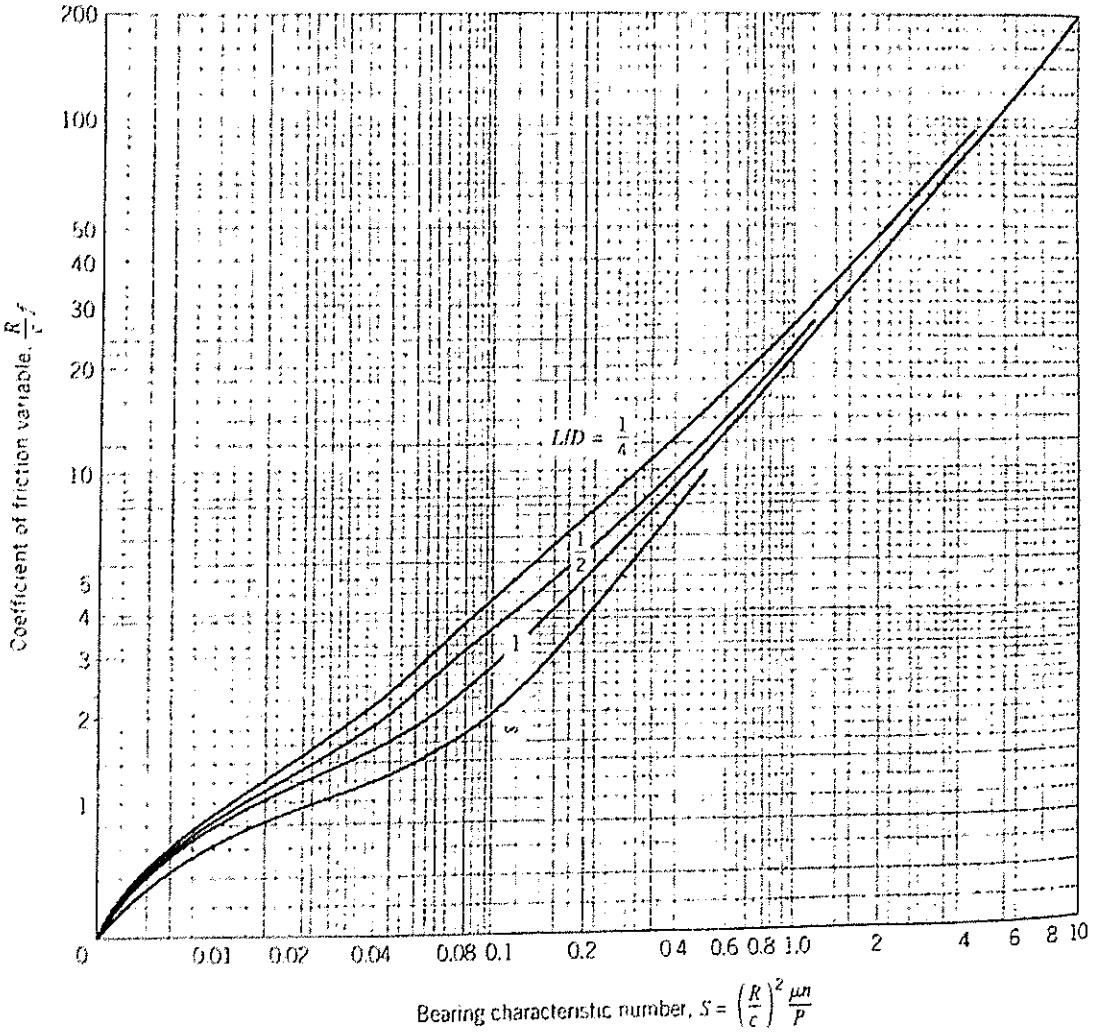

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Chart A6 – Maximum film pressure

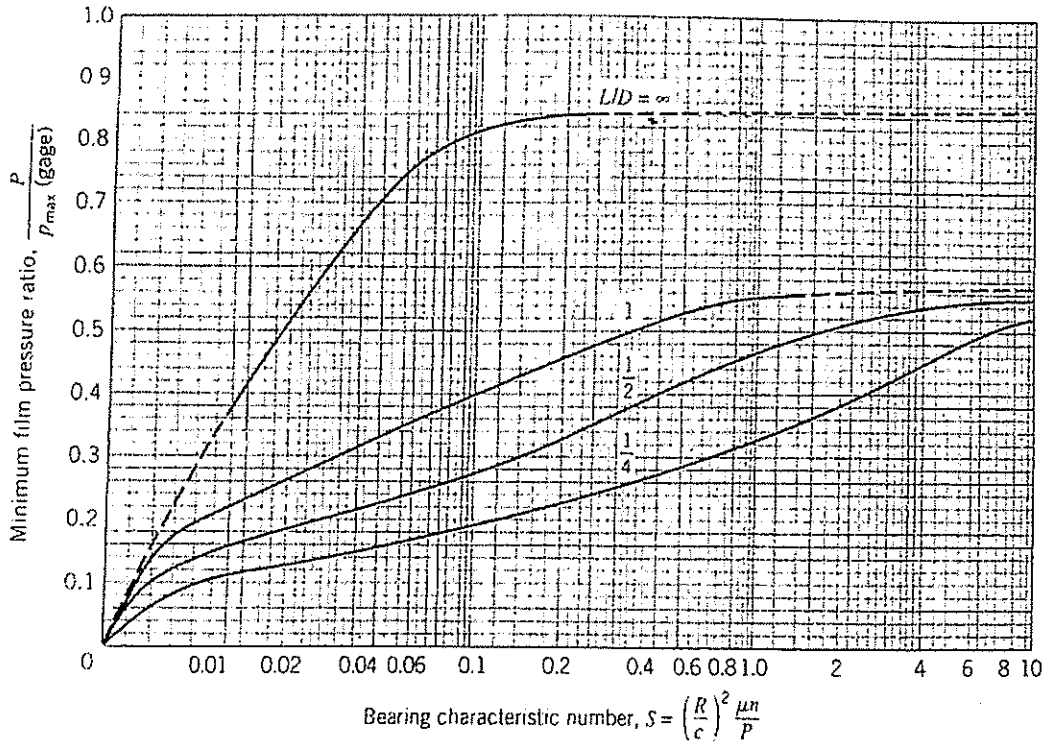

Chart A7 – Position of the minimum film thickness

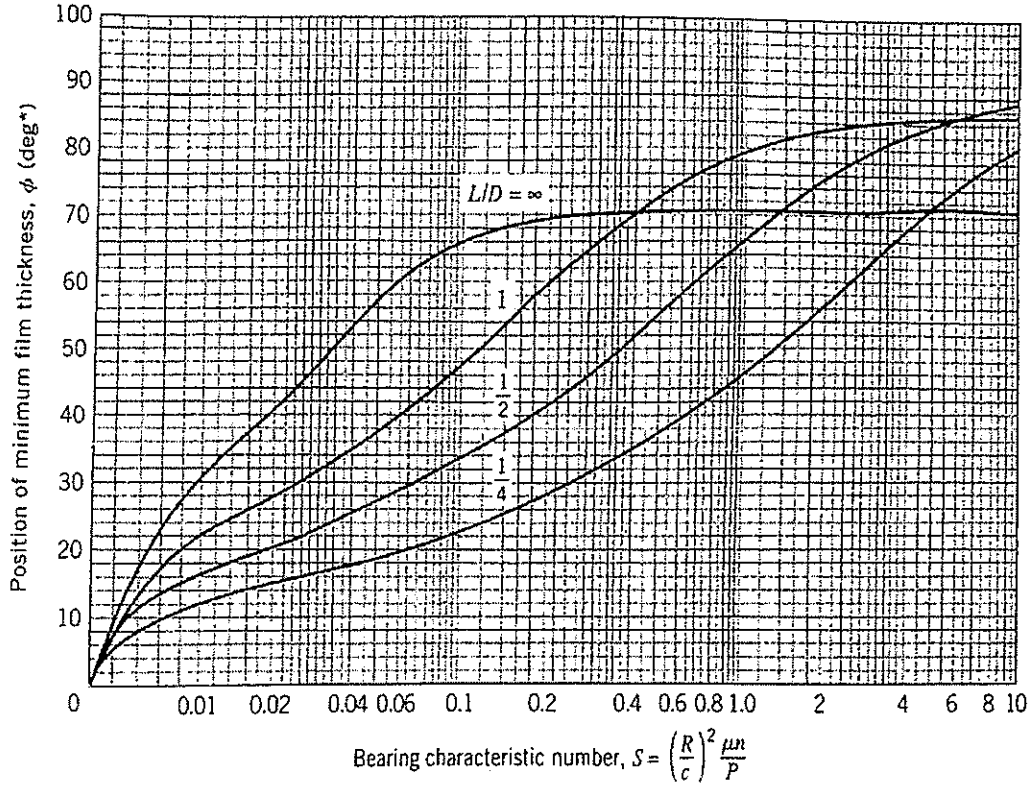

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Chart A8 – Positions of the maximum film pressure and film termination

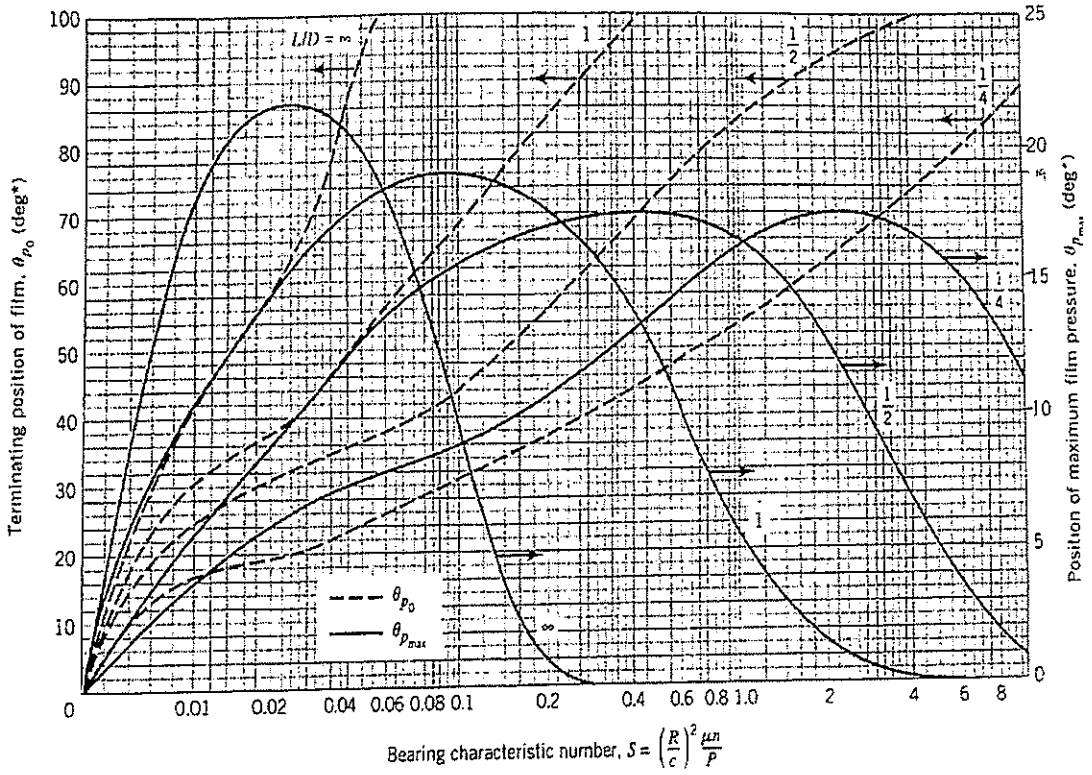
Terminating position of film, θ_{p_0} (deg*)
Position of maximum film pressure, $\theta_{p_{max}}$ (deg*)
Bearing characteristic number, $S = \left(\frac{R}{c}\right)^2 \frac{\mu n}{P}$
L/D = ∞
1
1/2
1/4
∞
θ_{p_0}
$\theta_{p_{max}}$
0 10 20 30 40 50 60 70 80 90 100
0 5 10 15 20 25
0.01 0.02 0.04 0.06 0.08 0.1 0.2 0.4 0.6 0.8 1.0 2 4 5 8

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Chart A9 – Flow variable

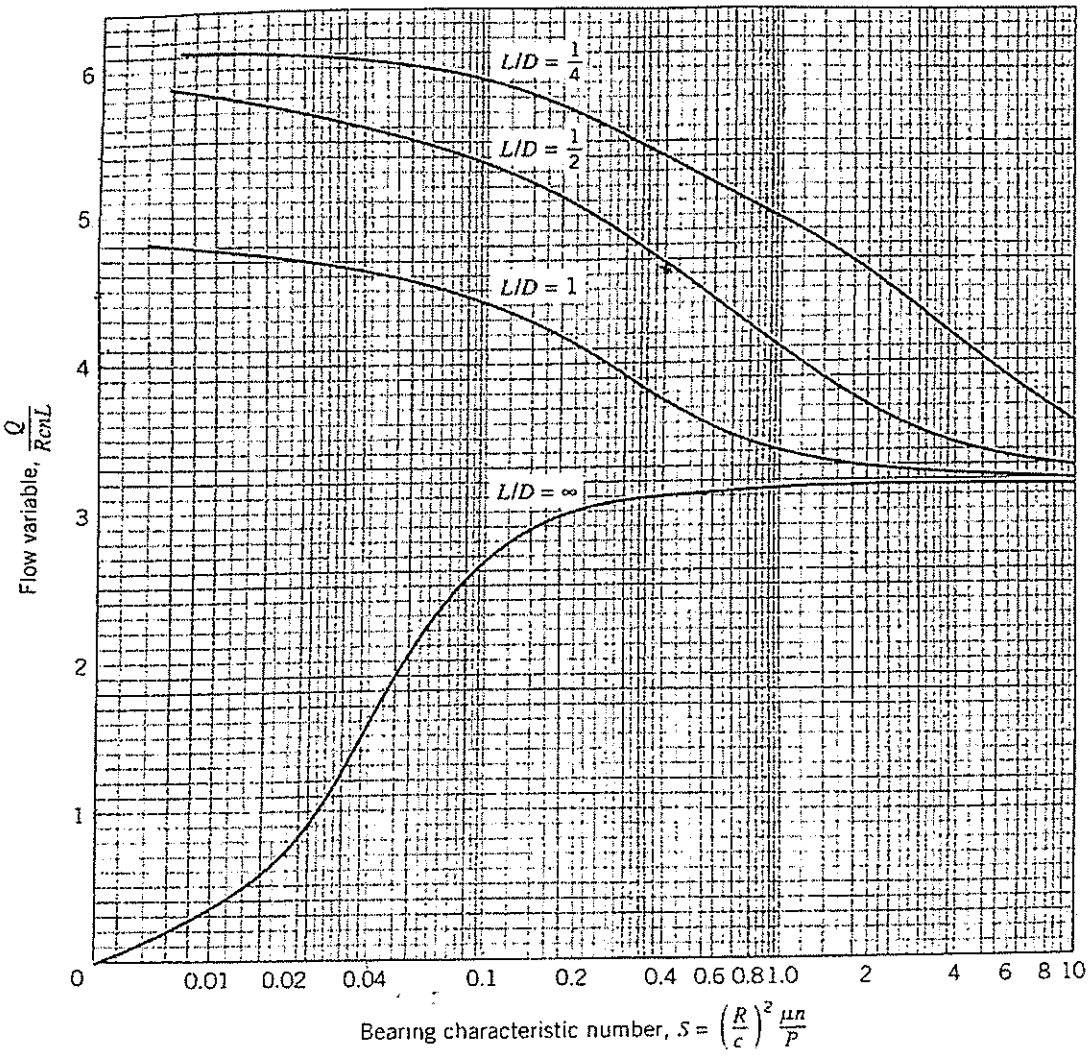

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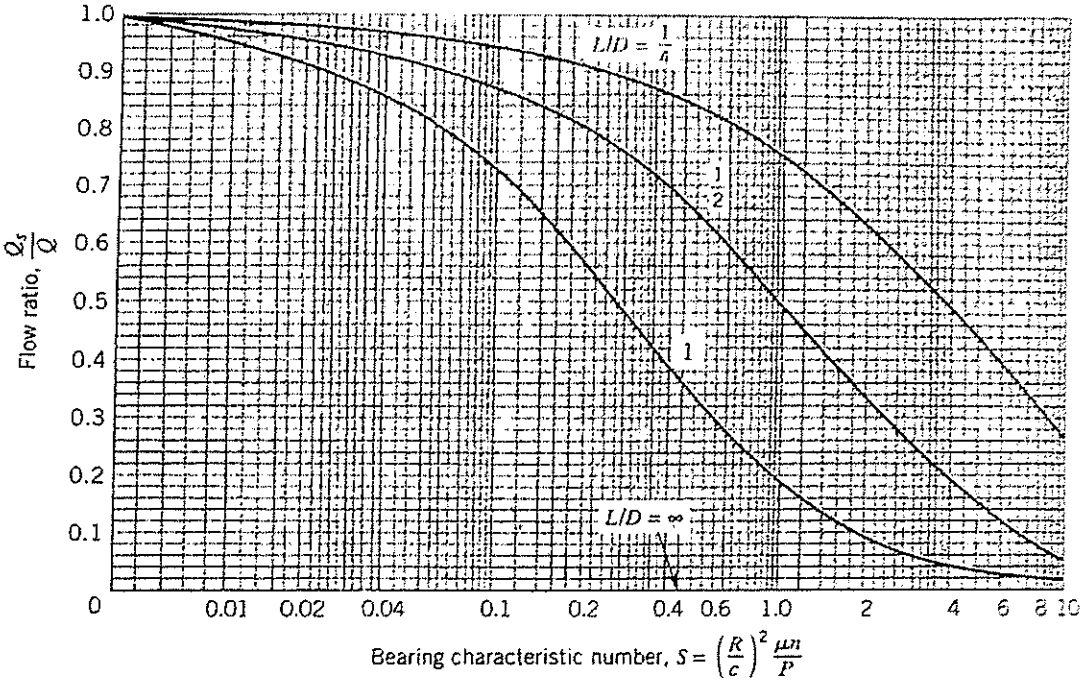
Chart A10 – Flow ratio



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